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(54) Fuel injection mechanical pump, in particular for internal combustion engines with controlled ignition.

(57) The present invention relates to a mechanical pump for the direct injection of fuel into the combustion chamber of an internal combustion engine, of the reflux with interrupted delivery type, wherein the plunger (14) delivers the fuel from a delivery chamber (16) into the combustion chamber up to a position in its stroke, at which the fuel reflows towards its tank through ducts (22,23,24,25) provided in the plunger (14); such ducts comprise at least two helical grooves (22) provided along the outer wall of the plunger (14), communicating with the delivery chamber (16) through outer longitudinal grooves (23) and through at least a longitudinal duct (24) provided inside the plunger, so as to accomplish a pump with large fuel reflux section area.

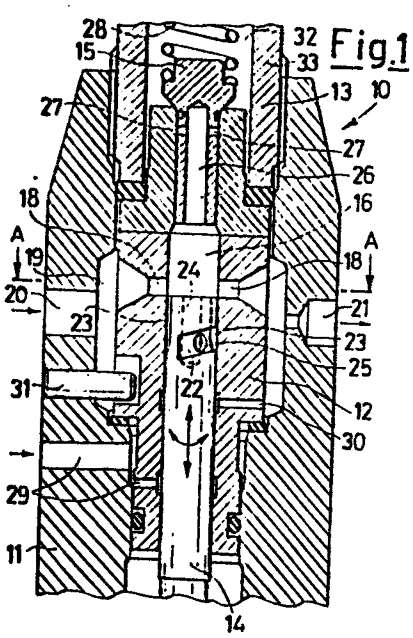

Fig.1

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The present invention relates to a mechanical pump for the injection of fuel intended for being applied in particular to internal combustion engines with controlled ignition.

The advantages are known of the use of the direct injection of fuel into the combustion chamber of the engines with controlled ignition.

Such advantages can be generally summarized as a lower tendency to the knocking by the motor, with the consequent possibility of the adoption of higher compression ratios with higher operating efficiencies, and as a quicker accomodation of the motor to a fast load change required by the environment.

In case of use of the direct injection in a controlled-ignition, two stroke cycle engine, to the aforementioned advantages, also the fact is added that by directly injecting the fuel into the combustion chamber with a suitable phasing, the loss of fuel mixture from the exhaust port during the scavenge stroke is avoided, with consequent beneficial effects on the consumption of fuel and on the emissions of unburnt hydrocarbons.

It is known in the specific art that in order to be able to carry out a correct fuel injection, i.e. atomized enough to guarantee a sufficient mixing during the short contact time with air, a high injection pressure (about 40 bar) is necessary. To the purpose of ensuring such an injection pressure, the use of a pump of mechanical type is necessary. Such a pump, analogous as for the particular components and operation to the pumps adopted in the diesel cycle engines, consists in a pumping element or plunger sliding with a notable precision inside a cylinder, with a reciprocating motion controlled by the outside by means of a cam-follower system, or by an eccentric, in a desmodromic way, driven by the same engine.

The governing of the fuel delivery takes place generally by diverting to the outside the exceeding delivery by putting in communication, through a groove provided on the outer surface of the plunger, the delivery chamber of the pump with the fuel tank through a reflux port, according to a commonly used technique.

However, to the contrary of what happens in the diesel cycle engines wherein, with operating in an excess of air, the power is function of the amount of injected fuel only (governing by quality), in the controlled-ignition engines the power is a function of the weight of fresh mixture (governing by quantity), in which however the air/fuel ratio must be kept substantially constant.

In all practical applications, the governing of the delivery of the pump is obtained by means of the rotation of the plunger around its axis; the plunger has indeed on its outer surface a helical groove, which allows the reflux of fuel and hence carries out the governing thereof by fractions of its total stroke, univocally depending on the angular position of the plunger inside the cylinder inside which it runs.

The theoretical delivery per plunger stroke would be constant, if blow-by Iosses inversely proportional to the square of the pumping rate and directly proportional to the delivery pressure, growing with the increasing of the pumping rate, did not exist. Such losses render the actual performance of the pump very variable as a function of its rotation rate (number of revolutions of the pump per time unit). Substantially, in a determined angular position of the plunger, the delivery per plunger stroke increases with increasing rotation rates according to a curve with decreasing derivative.

This pump delivery variation law does not fit to the requirement of fuel per engine revolution with the varying of the rotation rate of this latter (number of engine revolutions per time unit), intrinsic of the controlled ignition engines.

In fact, the necessary amount of fuel per engine revolution increases with the increasing of the engine rotation rate in accordance with an equal increase of the intaken air up to the point of optimum proportioning of the engine, i.e., to the point of maximum actual average pressure corresponding to the maximum driving torque; from this point on, the necessary amount of fuel per engine revolution decreases in accordance with an equal reduction of the intaken air.

The injection pump as described, on the contrary, would feed the motor with an increasing amount of fuel per engine revolution also beyond the point of maximum actual average pressure, so as to enrich too much the mixture and to cause a malfunctioning of the engine.

Apart governor devices have been proposed, which provide for the reduction of the amount of fuel per engine revolution at the proper time, i.e., after the point of maximum actual average pressure.

One type of such governor devices envisages a tridimensional cam, axially moved by a centrifuge device driven by the engine, and angularly moved by the element govern ing the air intake by the engine, which actuates an element governing the angular position of the plunger of the pump and hence governing the amount of fuel injected into the engine.

Such a governor device guarantees a perfect pump - engine coupling, but is extremely sophisticated, expensive bulky and hence, among others, not much suitable to the use on small engines, such as e.g. those with the two-stroke cycle, intended for light motorvehicles and motorscooters.

A second type of these governor devices, of pneumatic type, envisages the use of a membrane controlled by the suction pressure at the engine intake, which actuates the said pump plunger angular position governing device.

Such governor device is certainly simpler than the preceding one, but is less precise and is acceptable only for uses in semistationary engines, in that establishing a perfect correspondence between the suction pressure and the amount of intaken air is quite problematical.

Purpose of the present invention is hence to propose a mechanical injection pump for controlled-ignition engines, allowing an optimum coupling with the engine in terms of fuel feeding, avoiding the use of supplementary governor devices, and which is at the same time structurally simple.

Such purpose is achieved by means of a mechanical pump for the direct injection of fuel into the combustion chamber of an internal combustion engine, comprising a plunger, axially sliding inside a cylinder and rotating around its own axis, which delivers the fuel coming from a tank into a delivery chamber provided within the same cylinder towards a duct connected with the combustion chamber of the engine up to a position in the stroke of the plunger at which the fuel flows back, through ducts provided in the plunger, towards the tank, characterized in that said ducts comprise at least two helical grooves provided along the outer wall of the plunger, in communication with longitudinal grooves always provided along the outer wall of the plunger, and in communication moreover with at least a longitudinal duct provided in the plunger, said longitudinal grooves and said longitudinal duct putting the helical grooves in communication with the delivery chamber, in determined positions along the cylinder axis said helical grooves provided in the plunger being, with one of their portions, in correspondence of ports provided in the cylinder connected with the tank, for the flowing back of the fuel into the tank.

To the purpose of understanding the characteristics and the advantages of the present invention hereinunder an exemplifying embodiment thereof is disclosed, as illustrated in the attached drawing table wherein:

Fig. 1 shows a longitudinal section of a mechanical injection pump according to the invention;

Fig. 2 shows a detail of the pump of fig. 1;

Figs. 3, 4 and 5 show in a detailed way, the pump of fig. 1 in three different operating positions;

Figs. 6, 7 show in a detailed way, in cross section according to the path A-A, the pump of fig. 1 in two different operating positions.

The pump shown in fig. 1, generally indicated with 10, comprises a pump body 11 wherein a cylinder 12 and a valve body 13 are housed in a longitudinal connection relation ship.

Inside the cylinder 12 a plunger 14 is sliding.

Within the valve body 13 a shutter 15 aligned with the plunger 14 is housed.

Between the plunger 14 and the shutter 15 a delivery chamber 16 is defined, which communicates through two transversal passageways 18 provided in the cylinder 12, with an annular chamber 19. The annular chamber 19 communicates on one side, through a passageway 20, with a supply of pressurized fuel and on the other side, through a passageway 21, with a tank containing the same fuel.

On the outer wall of the plunger 14 two opposite to each other helical grooves 22 are provided, as well as two longitudinal grooves 23, they too being opposite to each other, each one of which intersecates at an end a respective helical groove 22 and ends at the other end in correspondence of the delivery chamber 16.

Inside the plunger 14 an axial duct 24 (well visible in fig. 2) is moreover provided, closed at an end, and terminating at its other end in correspondence of the delivery chamber 16. The axial duct 24 communicates with the two grooves 22 by means of two transversal bores 25 always provided in the plunger 14.

Within the shutter 15 an axial duct 26 is also provided, closed at an end, and terminating at its other end in correspondence of the delivery chamber 16. Within the shutter 15 two transversal through bores 27 are moreover provided, which put the axial duct 26 in communication with the outer of the shutter. The shutter is elastically pressed in closure position against a seat of the valve body 13 by a spring 28.

Within the pump 10 moreover ducts 29 for the inflow of pressured oil into the seat of the cylinder 12 within which the plunger 14 slides; a duct 30 for the recovery of the fuel leaking out between the said seat of the cylinder 12 and the same cylinder; and a stud 31 preventing the cylinder 12 from rotating inside the pump body 11 are provided.

The operating way of the disclosed pump 10, as applied to a controlled-ignition engine, not shown, is as follows.

In the position of fig. 1, the plunger 14 is in its position of lower dead point, and the fuel inflows through the passage 20, the annular chamber 19 and the passageways 18 into the delivery chamber 16. At the beginning of its stroke upwards, the plunger 14 causes the fuel to flow back into the annular chamber 19. As soon as the upper edge of the plunger 14 covers completely the ports of the passageways 18, as shown in fig. 3, the delivery geometrically starts (provided that, as in fig. 6, the longitudinal grooves 23 are not in correspondence of the ports of the passageways 18), i.e., the fuel pressurized inside the delivery chamber 16 causes the lifting and hence the opening of the shutter 15, counteracted by the spring 28; the bores 27 put the duct 26 in communication with an inner duct 32 of a delivery joint 33 connected with the engine's combustion chamber and hence the fuel outflows from the delivery chamber 16, through the duct 26, the bores 27 and the duct 32 into the said combustion chamber. In fig. 4 an intermediate position of the useful stroke of the piston is shown.

When the upper edge of the two helical grooves 22 reach the lower edges of the ports of the passageways 18, as shown in fig. 5, the delivery geometrically ends, in that the fuel pressurized inside the delivery chamber 16 flows back into the helical grooves 22, through the longitudinal grooves 23 and also through the axial duct 24 and the bores 25, and hence flows back, through the passageways 18, into the annular chamber 19. The plunger 14, subsequently, by having ended its stroke upwards, returns downwards, and at its lower dead point a new cycle begins.

The governing of the delivery of the pump 10 is obtained by rotating the plunger 14 relatively to the cylinder 12. In fact, by varying the positioning of the upper edge of the helical grooves 22 relatively to the ports of the passageways 18, the useful stroke of the plunger 14 varies, as known, and hence the delivery.

In the angular position of the plunger 14 in which the longitudinal grooves 23 are in correspondence of the ports of the passageways 18, as shown in fig. 7, a zero delivery is obtained, in that the fuel present inside the delivery chamber 16 outflows throughout the upwards stroke of the plunger 14 exactly through the longitudinal grooves 23 and the passageways 18 into the annular chamber 19.

In fig. 1 some arrows are shown, which indicate both the reciprocating and the rotary motion of the plunger 14, as well as the inlet of the fuel to and the outlet of it from the pump body 11, and the inlet of the oil as hereinabove mentioned.

The twin helical grooves 22 with their respective longitudinal grooves 23, the axial duct 24 and the bores 25 increase the reflux section area of the pump, so as to reduce the influence of the fuel leakage losses on the delivery law to a considerable extent; in practice, the curve of the delivery per plunger stroke as a function of the rotation rate of the pump, mentioned in the introduction, which has a growing outline with negative derivative in the known pumps, changes in the pump 10 in such a way that the curve is growing up to a certain point and then decreases, as it has been confirmed by experimental results. It is hence clear that it is possible to fit the variation law of the delivery of the pump 10 per stroke of the plunger 14, with the varying of the rotation rate of the pump, to the variation law of the amount of fuel per revolution of the controlled-ignition engine,with the varying of the rotation rate of the engine, which as it has been outlined in the introduction, envisages a growing curve up to the point of maximum actual average pressure, and decreasing after this point.

The increase of the reflux section area of the pump 10 leads also to advantages in the situation of zero delivery of the pump as mentioned above.

This favours indeed the reflux of all the fuel present inside the delivery chamber to a notable extent and the resistance to the reflux is thus notably reduced, in particular at the high rotation rates of the pump. In the known pumps, on the contrary, at the high pump rotation rates, this resistance to the reflux is so high that a fuel delivery is anyway obtained, even if of minimum value, with consequent increase in consumptions and environmental polluting substances.

The pump 10 is found to be particularly advantageous for applications to two-stroke engines which operate up to high rotation rates. It can be applied however to any internal combustion engines, whether of the controlled-ignition type or not, and hence also to a diesel engine, in those cases in which the increase of the reflux section area leads to technical advantages.

We underline that this is obtained with a very simple pump configuration, obtainable by means of elementary mechanical machining processes, and hence constructively cheap.

Of course, no supplementary devices are thus necessary to the purpose of correctly coupling the pump and the motor, in terms of feeding.

By operating on the geometry of the two grooves 22, whose upper edges may be made become operative at different heights, the delivery law of the pump 10 may be so adapted, as to fit to the intrinsic requirement of fuel by any engine types.

Different shapings of the pump as disclosed may be of course obtained.

More than two helical grooves with their respective longitudinal grooves, connection passageways to the reflux chamber (in the disclosed and illustrated example, the annular chamber 19), and connection bores to the axial duct of the plunger may be e.g. provided.

Instead of one single axial duct, more longitudinal ducts always provided inside the plunger may be provided, serving to the same function as of the axial duct, and, e.g., each one of them may be in communication with a respective helical groove.

Claims

1. Mechanical pump for the direct injection of fuel into the combustion chamber of an internal combustion engine, comprising a plunger axially sliding inside a cylinder and rotating around its own axis, delivering the fuel incoming from a tank into a delivery chamber provided within the said cylinder, towards a duct connected with the combustion chamber of the engine up to a position of the plunger's stroke in which the fuel flows back towards the tank through ducts provided in the plunger, characterized in that said ducts comprise at least two helical grooves provided along the outer wall of the plunger, communicating with longitudinal grooves always provided along the outer wall of the plunger and communicating moreover with at least one longitudinal duct provided inside the plunger, said longitudinal grooves and said longitudinal duct putting the helical grooves in communication with the delivery chamber, in determined positions along the cylinder axis said helical grooves provided in the plunger being, with one of their portions, in correspondence of ports provided in the cylinder connected with the tank, for the flowing back of the fuel into the tank.

2. Pump according to claim 1, wherein said longitudinal grooves provided in the plunger are so positioned, as to be, in determined angular positions of the plunger, in correspondence of said ports provided in the cylinder for the overall flow back of all the flowed-in fuel into the delivery chamber.

3. Pump according to claim 1 or 2, wherein an outer longitudinal groove per each helical groove is provided,

said grooves crossing each other.

4. Pump according to claim 1 or 2, wherein one single longitudinal duct centrally provided along the axis of the cylinder is envisaged.

5. Pump according to claim 4, wherein said axial duct provided in the plunger communicates with the helical grooves through bores provided inside the plunger, and terminates at the end of the plunger in correspondence of the delivery chamber.

6. Pump according to claim 3, wherein per each helical and longitudinal groove one of said cylinder ports is provided.

7. Pump according to claim 3, wherein two pairs of helical and longitudinal grooves positioned in opposite positions to each other are provided.

Fig.1

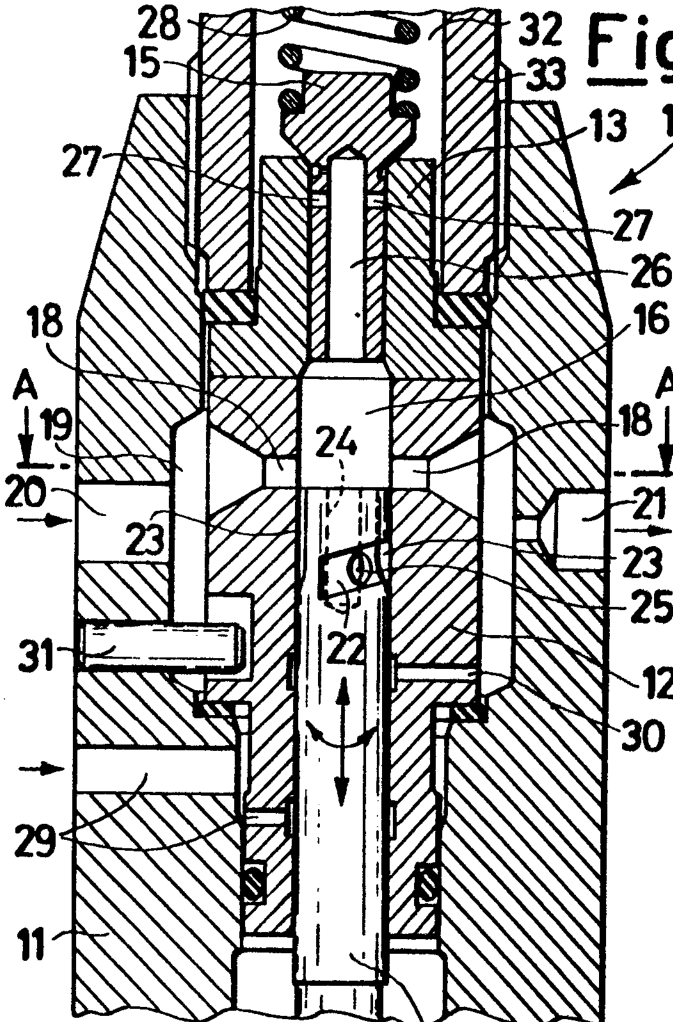


Fig.2

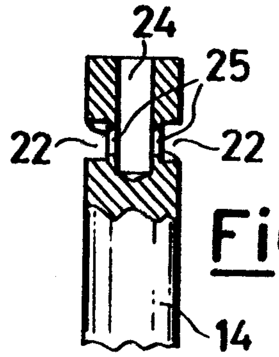


Fig.3

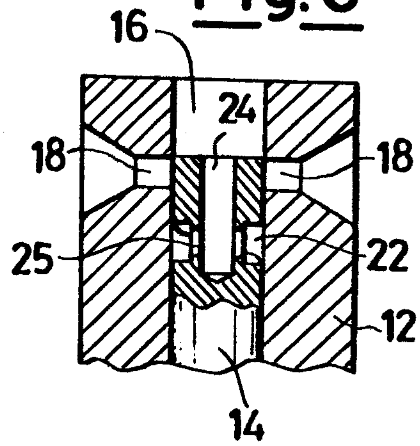


Fig.4

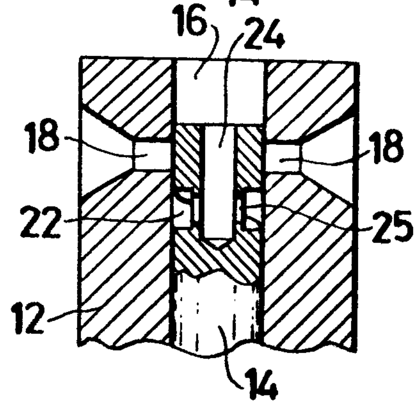


Fig.5

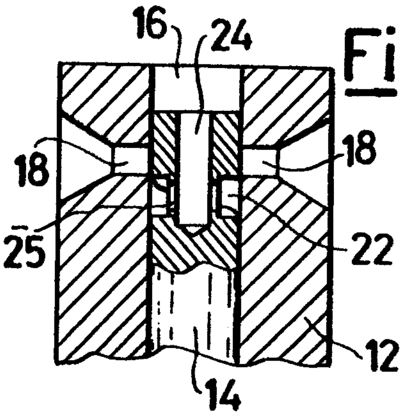


Fig.6

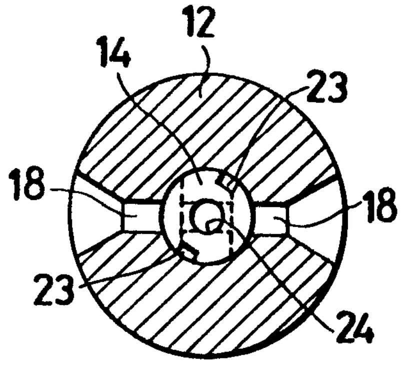
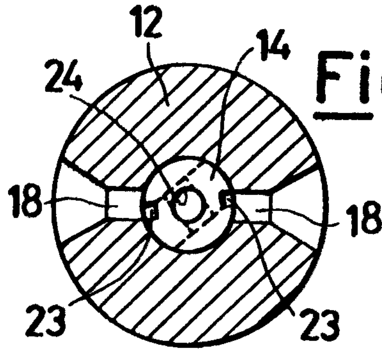


Fig.7



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EUROPEAN SEARCH REPORT

0209938

Application number

EP 86 20 1185

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int Cl 4)
X	US-A-2 429 806 (DESCHAMPS) * Column 1, lines 1-4; column 4, line 66 - column 5, line 15; column 11, line 44 - column 12, line 10; figures 14-19 *	1-7	F 02 M 59/26 F 02 M 69/02
	-----		**TECHNICAL FIELDS SEARCHED (Int Cl 4)**
			F 02 M

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	09-10-1986	HAKHVERDI M.

CATEGORY OF CITED DOCUMENTS

X : particularly relevant if taken alone
Y : particularly relevant if combined with another document of the same category
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